REMARKS

Amendments to the specification are reflected in the enclosed Substitute Specification and marked-up copy thereof. A reference to the PCT International Application and the Japanese priority application has been inserted. In addition, the sub-headings have been amended to conform to US format.

The claims have been amended as in the Article 34 Amendment filed during the International phase. In addition, "characterized in that" has been changed to --wherein-- and all multiple dependencies have been deleted.

Examination of this application in view of these amendments is respectfully requested. A copy of the International Preliminary Report on Patentability with amended claims is enclosed.

The Office is authorized to charge any underpayment or credit any overpayment to Kenyon & Kenyon's Deposit Account No. 11-0600.

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Respectfully submitted,

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SUBSTITUTE SPECIFICATION - MARKED-UP COPY

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~~DESCRIPTION~~

VARIABLE VALVE ACTUATION MECHANISM FOR AN INTERNAL COMBUSTION ENGINE

This is a 371 national phase application of PCT/JP2005/000949 filed 19 January 2005 claiming priority to Japanese Applications No. 2004-010896 filed 19 January 2004, No. 2004-153936 filed 24 may 2004, and No. 2004-226145 filed 02 August 2004, the contents of which are incorporated herein by reference.

~~TECHNICAL~~ FIELD OF THE INVENTION

The present invention relates to a variable valve actuation mechanism for an internal combustion engine that adjusts the actuation of valves by axially moving a control shaft with an actuator in the engine.

BACKGROUND [[ART]] OF THE INVENTION

For example, Japanese Laid-Open Patent Publication No. 2001-263015 discloses such a variable valve actuation mechanism. In this mechanism, a control shaft is axially moved with an actuator, so that a slider gear engaged with the control shaft is moved axially. Accordingly, the valve actuation such as the valve duration angle and the valve lift is adjusted.

~~DISCLOSURE~~ SUMMARY OF THE INVENTION

Since the control shaft of a variable valve actuation mechanism axially moves the slider gear, a portion of the control shaft that is engaged with the slider gear receives a great force. Particularly, when the slider gear is moved in a

direction for increasing the valve lift, the engaging portion is likely to receive a great force. Therefore, if the control shaft has an insufficient strength, the engaging portion can be deformed. To prevent such deformation, a control shaft is made of a high-strength material such as an iron based material.

However, in a case where the cylinder head of an engine is made of a light alloy material such as an aluminum alloy to reduce the weight of the engine, the thermal expansion coefficient of the cylinder head is significantly greater than that of the iron based material. Specifically, the thermal expansion coefficient of the cylinder head is about twice as great as that of the iron-based material. Thus, if an iron-based material is used for the control shaft of a variable valve actuation mechanism, the position of the engaging portion of the control shaft relative to the cylinder head is displaced between when the engine is cold and when the engine has been warmed up. Therefore, even if the actuator detects the amount of movement of the control shaft to control the movement amount of the control shaft, the axial displacement of the engaging portion due to a temperature change can hinder an accurate control of a valve actuation such as the valve lift.

If the control shaft is made of a light alloy material such as an aluminum alloy that is the same material of the cylinder head, the thermal expansion coefficient of the control shaft will be the same as that of the cylinder head, and the valve actuation control will be accurate. However, since a light alloy material such as an aluminum alloy is not as strong as an iron-based material, the engaging portion is likely to be deformed by a reaction force applied by the slider gear.

Accordingly, it is an objective of the present invention to provide a variable valve actuation mechanism for an internal combustion engine that performs a highly accurate valve actuation control, while maintaining the strength of a control shaft.

Means for achieving the above objectives and advantages thereof will now be discussed.

A variable valve actuation mechanism for an internal combustion engine according to one aspect of the present invention includes an intervening drive mechanism, a control shaft, and an actuator. The intervening drive mechanism transmits drive force from a cam provided in a cylinder head of the engine to a valve. The control shaft is engaged with a valve actuation controller provided in the intervening drive mechanism and moves the valve actuation controller in an axial direction to adjust the valve actuation. The actuator moves the control shaft in an axial direction to adjust the valve actuation. The mechanism is characterized in that the control shaft has an engaging portion that is engaged with the valve actuation controller and is made of a high strength material, wherein a remaining portion of the control shaft other than the engaging portion is made of a material that is different from the material of the engaging portion, such that the thermal expansion coefficient of the entire control shaft is made closer to the thermal expansion coefficient of the cylinder head.

The engaging portion of the control shaft is made of a high strength material, and the remaining portion other than the engaging portion is made of a material that is different from the material of the engaging portion, such that the thermal expansion coefficient of the entire control shaft is made closer to the thermal expansion coefficient of the

cylinder head. Since the engaging portion of the control shaft has a high strength, the engaging portion is prevented from being deformed. The remaining portion of the control shaft other than the engaging portion is not made of the same material as the engaging portion, but is made of a different material, such that the thermal expansion coefficient of the entire control shaft is made closer to the thermal expansion coefficient of the cylinder head. That is, since the strength of the remaining portion other than the engaging portion does not need to be as high as the strength of the engaging portion, a material for adjusting the thermal expansion coefficient of the entire control shaft is selected rather than a material for improving the strength, such that the thermal expansion coefficient of the entire control shaft is made closer to the thermal expansion coefficient of the cylinder head. Therefore, even if the ambient temperature changes, the axial position of the engaging portion is hardly displaced.

Accordingly, the variable valve actuation mechanism of the present invention is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that the cylinder head be made of a light alloy material, wherein the engaging portion of the control shaft is made of an iron based material, and the remaining portion of the control shaft is made of a light alloy material.

Specifically, if the cylinder head is made of a light alloy material, the engaging portion of the control shaft is made of an iron based material, and the remaining portion of the control shaft is made of a light alloy material. In this case, since the cylinder head and the remaining portion of

the control shaft other than the engaging portion are both made of a light alloy and have similar or the same thermal expansion coefficients, the thermal expansion coefficient of the entire control shaft can be made closer to the thermal expansion coefficient of the cylinder head even if the engaging portion of the control shaft is made of an iron based material.

Accordingly, the variable valve actuation mechanism is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that the light alloy material be an aluminum alloy material or a magnesium alloy material.

The light alloy material may be an aluminum alloy material or a magnesium alloy material, and since these have similar or the same thermal expansion coefficients, the thermal expansion coefficient of the entire control shaft can be made closer to the thermal expansion coefficient of the cylinder head even if the engaging portion of the control shaft is made of an iron based material. Accordingly, the variable valve actuation mechanism is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that the remaining portion of the control shaft other than the engaging portion be made of the same material as the material of the cylinder head.

The remaining portion of the control shaft other than the engaging portion may be made of the same material as the material of the cylinder head. Accordingly, the thermal expansion coefficient of the entire control shaft is made closer to the thermal expansion coefficient of the cylinder

head even if the engaging portion is made of a high strength material such as an iron based material. Accordingly, the variable valve actuation mechanism is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that the material and the length of the engaging portion and the material and the length of the remaining portion of the control valve other than the engaging portion be set such that the thermal expansion coefficient of the control shaft is substantially the same as the thermal expansion coefficient of the cylinder head.

The material and the length of the engaging portion and the material and the length of the remaining portion of the control valve other than the engaging portion may be set such that the thermal expansion coefficient of the control shaft is substantially the same as the thermal expansion coefficient of the cylinder head. By designing the control shaft such that its thermal expansion coefficient is compatible to that of the cylinder head, the thermal expansion coefficient of the entire control shaft is made closer to and is substantially the same as the thermal expansion coefficient of the cylinder head. Accordingly, the variable valve actuation mechanism is capable of performing even more accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that the thermal expansion coefficient of the cylinder head be greater than that of the engaging portion, wherein the intervening drive mechanism is one of a plurality of intervening drive mechanisms each provided for one of the cylinders, the control shaft is common to all the intervening drive mechanisms, wherein the thermal expansion coefficient of the remaining portion of the control shaft

other than the engaging portion is greater than the thermal expansion coefficient of the engaging portion, wherein the relationship of the length of the engaging portion and the remaining portion other than the engaging portion is set such that, between each adjacent pair of the intervening drive mechanisms, the thermal expansion coefficient of the control shaft is lower than the thermal expansion coefficient of the cylinder head, and wherein the ratio of the length of the remaining portion other than the engaging portion to the length of the engaging portion between each adjacent pair of the intervening drive mechanisms gradually increases as the distance from the actuator to the pair increases.

The material used for the remaining portion other than the engaging portion is selected for a great thermal expansion coefficient, and therefore, in some cases, cannot be elongated due to an expected reduction of strength of expected increase in the cost. Therefore, instead of setting the thermal expansion coefficient for substantially or completely eliminating axial displacement of the engaging portion, it is advantageous for increasing the strength and reducing the cost to set a permissible range of axial displacement and to minimize the length of the remaining portion other than the engaging portion.

However, in sections closer to the actuator, shortening the remaining portion other than the engaging g portion causes axial displacement of the engaging portion to accumulate as the distance from the actuator increases. Therefore, if the ratio of the length of the remaining portion other than the engaging portion and the engaging portion is the same even if the distance from the actuator increases, the axial displacement of the engaging portion can exceed the permissible range.

In the present invention, the ratio of the length of the remaining portion other than the engaging portion to the length of the engaging portion is gradually increased as the distance from the actuator increases. This prevents each engaging portion from being displaced beyond the permissible range. Also, the strength of the control shaft is prevented from being degraded, or the costs of the control shaft are prevented from being increased.

It is preferable that the intervening drive mechanisms be arranged substantially at a constant interval, and that the length of the remaining portion other than the engaging portion between each adjacent pair of the intervening drive mechanisms increase as the distance from the actuator to the pair increases.

If the intervening drive mechanisms are arranged at a constant interval, the length of the remaining portion other than the engaging portion is set to increase as the distance from the actuator to the pair increases. Accordingly, the axial displacement of all the engaging portions is prevented from exceeding the permissible range. Also, the strength of the control shaft is prevented from deteriorating, and the cost of the control shaft is prevented from being increased.

It is preferable that the engaging portions of the control shaft and the remaining portions of the control shaft other than the engaging portions be formed separately and arranged along a common axis while being brought into contact with one another to form the control shaft, and that the actuator be provided at one end of the control shaft and urging means be located at the other end of the control shaft to urge the control shaft toward the actuator.

The engaging portions of the control shaft and the

remaining portions of the control shaft other than the engaging portions do not need to be formed integrally, but may be formed separately and located between the actuator and the urging means. This always permits the control shaft to be moved axially while the engaging portions and the remaining portions other than the engaging portions are in a contacting state when the actuator operates. Since the engaging portions and the remaining portions other than the engaging portions, which are made of different materials, do not need to be connected, the manufacture of the control shaft is simplified.

It is preferable that the continuity of the material of the remaining portion other than the engaging portion along the axial direction of the control shaft be maintained.

In this manner, the material of the remaining portion other than the engaging portion along the axial direction of the control shaft is continuously arranged. That is, the material of the remaining portion other than the engaging portion is not separated in the axial direction by the engaging portion made of a different material. Therefore, in the axial direction, the thermal expansion coefficient of the material of the remaining portion other than the engaging portion is dominant in the control shaft. Therefore, even if the thermal expansion coefficient of the engaging portion is not considered and only the strength of the engaging portion is considered, the thermal expansion coefficient along the axis of the control shaft is hardly influenced.

Therefore, a high strength material is selected for the engaging portion, and a material having substantially the same thermal expansion coefficient as that of the cylinder head is selected for the remaining portion other than the engaging portion. Accordingly, while maintaining the strength of the control shaft, it is extremely easy to form a control

shaft having a thermal expansion coefficient that is extremely close to or the same as that of the cylinder head.

It is preferable that the remaining portion other than the engaging portion be formed integrally, and that the engaging portion be buried in and supported by the remaining portion.

In this manner, the remaining portion other than the engaging portion is formed integrally, and the engaging portion is buried in and supported by the remaining portion. Thus, the thermal expansion coefficient of the material of the remaining portion other than the engaging portion is dominant in the axial direction of the control shaft.

It is preferable that the engaging portion be engaged with the valve actuation controller by means of a control pin, and that the engaging portion be provided about the control pin in the control shaft to support the control pin.

Since the engaging portion is provided only about the control pin to support the control pin, the area on the control shaft occupied by the engaging portion is limited to a certain extent.

Therefore, in the axial direction of the control shaft, the thermal expansion coefficient of the material of the remaining portion other than the engaging portion can be easily made dominant.

It is preferable that the valve actuation controller be engaged with a control pin supported by the engaging portion and moves as the control shaft moves in the axial direction, and that the intervening drive mechanism include an input portion and an output portion, wherein the input portion is

engaged with the valve actuation controller by means of a first spline mechanism to receive valve drive force from the cam, and transmits the valve drive force to the valve actuation controller, wherein the output portion is engaged with the valve actuation controller by means of a second spline mechanism to receive the valve drive force from the valve actuation controller, and transmits the valve drive force to the valve, and wherein the helix angle of the first spline mechanism is different from the helix angle of the second spline mechanism, so that, as the control shaft moves axially, the relative positions of the input portion and the output portion are changed and the valve actuation is adjusted.

The intervening drive mechanism may be configured as described above, and the valve actuation can be adjusted by driving the control shaft. In the variable valve actuation mechanism having such an intervening drive mechanism, the control shaft having the above described structure is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

A variable valve actuation mechanism for an internal combustion engine, which has a plurality of cylinders, according to another aspect of the present invention includes intervening drive mechanisms each provided for one of the cylinders, a control shaft, and an actuator. Each intervening drive mechanism transmits drive force from one of cams provided in a cylinder head of the engine to a valve. The control shaft is engaged with a valve actuation controller provided in each intervening drive mechanism and moves the valve actuation controllers in an axial direction to adjust the valve actuation. The actuator moves the control shaft in an axial direction to adjust the valve actuation. The mechanism is characterized in that the valve clearance of

each valve is adjusted by a lash adjuster, wherein the leak down property of the lash adjusters are set different among the cylinders to suppress variation of the valve actuation among the cylinders due to a difference in the thermal expansion coefficient between the control shaft and the cylinder head in relation to the thermal expansion coefficient of each intervening drive mechanism.

Each lash adjuster has a leak down property, which refers to a property in which when receiving a certain load, the lash adjuster is moved downward due to oil leakage. If the leak down amount is small, the valve duration angle and lift corresponding to the amount of valve actuation transmitted from the intervening drive mechanisms are great. As the leak down property amount increases, the valve duration angle and lift for the same valve actuation value are reduced.

The sensitivity to the valve actuation changes due to the leak down properties is related to the viscosity of hydraulic oil. That is, if the viscosity of the hydraulic oil is high, the amount of oil leakage is reduced, which lowers the sensitivity of valve actuation changes due to the leak down properties. That is, even if the leak down properties greatly vary among the lash adjusters, a high viscosity of the hydraulic oil prevents the difference in the leak down properties from varying the valve actuation. Also, a low viscosity of the hydraulic oil permits the difference in the leak down properties to vary the valve actuation by a greater degree.

In the present embodiment, the leak down properties are varied among the cylinders to suppress variation of the valve actuation due to the difference in the thermal expansion coefficient of the control shaft and the cylinder head in

relation to the intervening drive mechanisms each provided for one of the cylinders. Therefore, the differences in the thermal expansion coefficient do not cause any problems when the engine has been warmed up.

Further, when the engine is cold, that is, when there is no variation in the valve actuation due to differences in the thermal expansion coefficient, the hydraulic oil has a high viscosity due to a low temperature. Therefore, even if the leak down properties vary among cylinders, the valve actuation is hardly varied by the differences in the leak down properties.

Therefore, even if the thermal expansion coefficients are different between the control shaft and the cylinder head, the valve actuation is prevented from being varied over the entire temperature range of the engine.

Accordingly, the variable valve actuation mechanism of the present invention is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that, by creating the difference in the thermal expansion coefficient, the leak down property value of the lash adjuster provided for a cylinder in which the valve actuation value is relatively increased due to a high temperature be set greater than the leak down property of a cylinder in which the valve actuation value is relatively decreased due to a high temperature.

More specifically, by providing a lash adjuster having leak down property for each cylinder, a difference in the leak down property is prevented from influencing the valve actuation when the engine is cold. Also, when the engine has

been warmed up, a difference in the leak down property permits the cylinders to have the same leak property.

Accordingly, the variable valve actuation mechanism of the present invention is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

A variable valve actuation mechanism for an internal combustion engine, which has a plurality of cylinders, according to another aspect of the present invention includes intervening drive mechanisms each provided for one of the cylinders, a control shaft, and an actuator. Each intervening drive mechanism transmits drive force from one of cams provided in a cylinder head of the engine to a valve. The control shaft is engaged with a valve actuation controller provided in each intervening drive mechanism and moves the valve actuation controllers in an axial direction to adjust the valve actuation. The actuator moves the control shaft in an axial direction to adjust the valve actuation. The variable valve actuation mechanism is characterized in that the valve clearance of each valve is adjusted by a lash adjuster, wherein the pressure of oil supplied to the lash adjusters is independently adjusted for each cylinder according to the temperature of the internal combustion engine to suppress variation of the valve actuation among the cylinders due to a difference in the thermal expansion coefficient between the control shaft and the cylinder head in relation to the thermal expansion coefficient of each intervening drive mechanism.

Increasing the pressure of oil supplied to the lash adjusters decreases the amount of leak down. Accordingly, the valve actuation is increased. Decreasing the supplied oil pressure increases the amount of leak down. Accordingly, the

valve actuation is reduced. Thus, by independently adjusting the pressure of oil supplied to the lash adjusters for each cylinder according to the temperature of the internal combustion engine, variation of the valve actuation among the cylinders due to a difference in the thermal expansion coefficient between the control shaft and the cylinder head in relation to the thermal expansion coefficient of the intervening drive mechanisms each provided for one of the cylinders is suppressed over the entire temperature range of the engine.

Accordingly, the variable valve actuation mechanism of the present invention is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

It is preferable that, according to the temperature of the internal combustion engine, the pressure of oil supplied to the lash adjuster provided for a cylinder in which the valve actuation value is relatively increased due to a high temperature be set smaller than the pressure of oil supplied to a cylinder in which the valve actuation value is relatively decreased due to a high temperature.

More specifically, by adjusting the pressure oil supplied to the lash adjusters each provided for one of the cylinders, variation of the valve actuation values among the cylinders, that is, variation of the valve duration angles and lifts, are suppressed both in a low temperature state and a high temperature state.

Accordingly, the variable valve actuation mechanism of the present invention is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a cross-sectional view illustrating an engine and a variable valve actuation mechanism according to a first embodiment;

Fig. 2 is a plan view illustrating the engine shown in Fig. 1;

Fig. 3 is a perspective view illustrating an intervening drive mechanism used in the variable valve actuation mechanism shown in Fig. 1;

Figs. 4(A) and 4(B) are perspective views, with a part cut away, illustrating the intervening drive mechanism shown in Fig. 3;

Fig. 5 is an exploded perspective view illustrating the intervening drive mechanism shown in Fig. 3;

Figs. 6(A) and 6(B) are perspective views, with a part cut away, illustrating an outer portion of the intervening drive mechanism shown in Fig. 3;

Figs. 7(A), 7(B), and 7(C) are diagrams illustrating a slider gear provided in the intervening drive mechanism shown in Fig. 3;

Fig. 8 is a perspective view illustrating the slider gear shown in Figs. 7(A) to 7(C);

Fig. 9 is a perspective cross-sectional view illustrating the slider gear shown in Figs. 7(A) to 7(C) taken along the axis;

Figs. 10(A), 10(B), and 10(C) are perspective views illustrating a support pipe and a control shaft provided in the slider gear shown in Figs. 7(A) to 7(C);

Fig. 11 is a perspective view illustrating the entire control shaft shown in Figs. 10(A) to 10(C);

Fig. 12 is a perspective view, with a part cut away, illustrating the intervening drive mechanism shown in Fig. 3;

Figs. 13(A) and 13(B) are diagrams showing operation of

the intervening drive mechanism shown in Fig. 3;

Figs. 14(A) and 14(B) are diagrams showing operation of the intervening drive mechanism shown in Fig. 3;

Fig. 15 is a cross-sectional view illustrating a stressed state of the intervening drive mechanism shown in Fig. 3 during operation;

Fig. 16 is a diagram illustrating a control shaft according to a second embodiment;

Fig. 17 is a graph showing a setting of the length of the control shaft shown in Fig. 16;

Figs. 18(A) and 18(B) are diagrams illustrating a control shaft according to a third embodiment;

Fig. 19 is a cross-sectional view illustrating the control shaft according to the third embodiment;

Fig. 20 is a diagram illustrating a control shaft according to a fourth embodiment;

Fig. 21 is a perspective view illustrating a variable valve actuation mechanism according to the fourth embodiment;

Fig. 22 is a graph showing leak down properties of the lash adjusters according to the fourth embodiment;

Figs. 23(A) and 23(B) are graphs showing the amounts of increase in the valve duration angle and the valve lift due to differences in thermal expansion coefficient, and the amounts of decrease in the valve duration angle and the valve lift due to leak down while the engine is cold according to the fourth embodiment;

Fig. 24 is a graph showing the relationship between an adjustment amount by a slide actuator in each cylinder and the valve duration angle and lift while the engine is cold according to the fourth embodiment;

Figs. 25(A) and 25(B) are graphs showing the amounts of increase in the valve duration angle and the valve lift due to differences in thermal expansion coefficient, and the amounts of decrease in the valve duration angle and the valve lift due to leak down after the engine has been warmed up

according to the fourth embodiment;

Fig. 26 is a graph showing the relationship between an adjustment amount by the slide actuator in each cylinder and the valve duration angle and lift after the engine has been warmed up according to the fourth embodiment;

Fig. 27 is a graph showing the relationship between an adjustment amount by a slide actuator in each cylinder and valve duration angle and lift after the engine has been warmed up according to a comparison example;

Fig. 28 is a perspective view illustrating a variable valve actuation mechanism according to the fifth embodiment;

Fig. 29 is a graph showing the relationship between the amount of decrease in the valve duration angle and the valve lift due to hydraulic pressures and leak down in lash adjusters according to the fifth embodiment;

Fig. 30 is a map for controlling the hydraulic pressure according to the fifth embodiment;

Fig. 31 is a perspective view illustrating a control shaft according to a sixth embodiment;

Fig. 32 is a graph showing changes in ratios of coupler shafts and axial displacements according to the sixth embodiment; and

Fig. 33 is a diagram, with a part cut away, illustrating a control shaft according to another embodiment.

~~BEST MODE FOR CARRYING OUT THE INVENTION~~ DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[First embodiment]

Fig. 1 illustrates the configuration of a variable valve actuation mechanism of an internal combustion engine, which is a gasoline engine 2 (hereinafter, simply referred to as an engine) in this embodiment. Fig. 1 is a vertical cross-sectional view at a cylinder. Fig. 2 is a plan view that

mainly shows the upper portion of the engine 2.

The engine 2 is mounted on a vehicle to drive the vehicle. The engine 2 has a cylinder block 4, pistons 6, and a cylinder head 8 placed on the cylinder block 4. The cylinder block 4, the pistons 6, and the cylinder head 8 of the engine 2 are made of a light alloy material, which is an aluminum alloy material.

The cylinder block 4 has cylinders 2a, the number of which is four in this embodiment. In each cylinder 2a, a combustion chamber 10 is defined by the cylinder block 4, the associated piston 6, and the cylinder head 8. Two intake valves 12 and two exhaust valves 16 are provided in each combustion chamber 10. The intake valves 12 open and close intake ports 14, and the exhaust valves 16 open and close exhaust ports 18.

The intake ports 14 of each cylinder 2a are connected to a surge tank through an intake passage formed in an intake manifold. Air is supplied to the cylinders 2a from the surge tank through an air cleaner and the intake ports 14. A fuel injector is provided in each intake passage to inject fuel toward the intake ports 14 of each cylinder 2a. In this embodiment, the present invention is applied to an engine where fuel is injected into a section upstream of the intake valves 12. However, the present invention may be applied to an in-cylinder fuel injection type gasoline engine, in which fuel is directly injected into each combustion chamber 10.

In this embodiment, since the intake air amount is adjusted by changing the valve lift of the intake valves 12, no throttle valve is provided in a section of the intake passage upstream of the surge tank. However, an auxiliary throttle valve may be provided. In a case where such an

auxiliary valve is provided, the auxiliary throttle valve is fully opened when cranking the engine 2, and is fully closed when stopping the engine 2. When, for example, an intervening drive mechanism 120 (described below) malfunctions, the opening degree of the auxiliary throttle valve is adjusted to control the intake air amount.

The intervening drive mechanisms 120 and roller rocker arms 52 are provided in the cylinder head 8. An intake camshaft 45 has intake cams 45a, each corresponding to one of the cylinders 2a. Each pair of the intake valves 12 are lifted by valve driving force transmitted to the intake valves 12 from the corresponding intake cam 45a through the intervening drive mechanism 120 and the corresponding roller rocker arms 52. At the transmission of valve driving force, a control shaft 132 is moved axially by the equilibrium between forces applied by a slide actuator 100 and an urging mechanism 102. Accordingly, the force transmitting state of the intervening drive mechanisms 120 is adjusted to vary the valve lift. The intake camshaft 45 is coupled to and rotated by a crankshaft 49 of the engine 2 with a timing sprocket (alternatively, a timing gear and a timing pulley) provided at an end and a timing chain 47.

An exhaust camshaft 46 is provided to be rotated by rotation of the engine 2. The exhaust camshaft 46 has exhaust cams 46a. The exhaust valves 16 of each cylinder 2a are opened and closed with a certain valve lift by the corresponding exhaust cams 46a by means of roller rocker arms 54. The exhaust ports 18 of each cylinder 2a are connected to an exhaust manifold, so that exhaust gas is emitted to the outside through catalytic converter.

An electronic control unit (hereinafter, referred to as ECU) 60 is constructed with a digital computer as the

dominant constituent, and includes a central processing unit (CPU), read only memory (ROM), random access memory (RAM), various driver circuits, an input port, and an output port, which are interconnected by a bidirectional bus. Through the input port, the ECU 60 receives signals representing an acceleration depression degree ACCP, an engine rotational speed NE, an intake air amount GA, an engine coolant temperature THW, an air-fuel ratio AF, and a reference crank angle G2.

Further, in this embodiment, the slide actuator 100 includes a ball screw shaft 100e. The axial position of the ball screw shaft 100e is detected by a shaft position sensor 100d. The ECU 60 receives a shaft position signal SL representing the axial position of the ball screw shaft 100e from the shaft position sensor 100d.

The output port of the ECU 60 is connected to the fuel injectors through the driver circuits. According to the operating state of the engine 2, the ECU 60 controls the fuel injectors to open, thereby performing fuel injection timing control and fuel injection amount control. The ECU 60 also performs other control procedures such as ignition timing control.

In this embodiment, the ECU 60 sends a drive signal Ds to a slide actuator driver circuit 62, thereby adjusting the axial position of the control shaft 132 through the slide actuator 100. In this manner, the ECU 60 controls the valve lift of the intake valves 12 to a target value.

The slide actuator 100 includes a motor 100a, a gear portion 100b, and a ball screw portion 100c. The rotation direction and the rotation amount of the motor 100a are adjusted by electricity from the actuator driver circuit 62.

The rotation is reduced by the gear portion 100b and transmitted to the ball screw portion 100c. Accordingly, the ball screw shaft 100e, which transmits an axial force to the control shaft 132, is moved in an axial direction corresponding to the rotation direction of the motor 100a by an amount corresponding to the rotation amount of the motor 100a.

The ECU 60 adjusts the rotation direction and the rotation amount of the motor 100a with the drive signal Ds such that the axial position of the ball screw shaft 100e detected by the shaft position sensor 100d corresponds to a target valve lift that is set in accordance with the operating state of the engine 2. Accordingly, the intake air amount is adjusted.

The intervening drive mechanisms 120 will now be described with reference to Figs 3 to 6.

Each intervening drive mechanism 120 includes an input portion 122 located in the center, a first rocker cam 124 provided at one end of the input portion 122, a second rocker cam 126 provided at the other end of the input portion 122, and a slider gear 128 located inside.

The input portion 122 includes a housing 122a, in which an axially extending space is defined. A right-hand helical spline 122b is formed in the inner circumferential surface of the housing 122a. Two parallel arms 122c, 122d project from the outer circumferential surface of the housing 122a. A shaft 122e extends parallel to the axis of the housing 122a between distal portions of the arms 122c, 122d. A roller 122f is rotatably attached to the shaft 122e. As shown in Fig. 1, force of a spring 122g is applied to the arms 122c, 122d or the housing 122a of each roller 122f. This causes the roller

122f to constantly contact the corresponding intake cam 45a. Each spring 122g is located between one of the input portions 122 and the cylinder head 8 or between one of the input portions 122 and a support pipe 130.

Each first rocker cam 124 includes a housing 124a, in which an axially extending space is defined. A left-hand helical spline 124b is formed in the inner circumferential surface of the housing 124a. An end of the housing 124a is covered with a ring-shaped bearing portion 124c. A small hole is formed in the center of the bearing portion 124c. A substantially triangular nose 124d projects from the outer circumferential surface of the housing 124a. The nose 124d forms a cam surface 124e with a concaved side.

The second rocker cam 126 includes a housing 126a, in which an axially extending space is defined. A left-hand helical spline 126b is formed in the inner circumferential surface of the inner space. An end of the housing 126a is covered with a ring-shaped bearing portion 126c. A small hole is formed in the center of the bearing portion 126c. A substantially triangular nose 126d projects from the outer circumferential surface of the housing 126a. The nose 126d forms a cam surface 126e with a concaved side.

As shown in Fig. 5, the first and second rocker cams 124, 126 are coaxially provided on the sides of the input portion 122 to contact the input portion 122 with the bearing portions 124c, 126c placed at the outer ends. Therefore, the first rocker cam 124, the second rocker cam 126, and the input portion 122 form a cylinder with an inner space.

In the inner space defined by each input portion 122 and the corresponding two rocker cams 124, 126 is located one of the slider gears 128 shown in Figs 7 to 9.

Each slider gear 128 is substantially cylindrical and has a right-hand input helical spline 128a is formed in a middle section of the outer circumferential surface. A first left hand output helical spline 128c is formed at one end of the slider gear 128. The first output helical spline 128c is spaced from the input helical spline 128a with a small diameter portion 128b in between. At the opposite side with respect to the first output helical spline 128c, a second left hand output helical spline 128e is formed. The second output helical spline 128e is spaced from the input helical spline 128a with a small diameter portion 128d in between. The outer diameters of the output helical splines 128c, 128e are smaller than the outer diameter of the input helical spline 128a.

An axial through hole 128f is formed in the slider gear 128. A circumferential groove 128g is formed in the inner circumferential surface of the through hole 128f at position corresponding to the input helical spline 128a. A pin insertion hole 128h is formed in the circumferential groove 128g. The pin insertion hole 128h extends radially and communicates with the outside.

The support pipe 130, which is partially shown in Fig. 10(A), is provided in the through hole 128f of each slider gear 128. The pipe 130 is slidable in the circumferential direction. As shown in Fig. 2, the support pipe 130 is common to all the intervening drive mechanisms 120. Elliptical holes 130a elongated in the axial direction are formed in the support pipe 130. Each elliptical hole 130a corresponds to one of the intervening drive mechanisms 120.

The control shaft 132, which is shown in Fig. 10(B), is provided in the support pipe 130 as shown in Fig. 10(C). The

control shaft 132 is slidable along the axial direction. As shown in Fig. 11, the control shaft 132 includes four engaging portions 132c and four coupler shafts 132d. The engaging portions 132c and the coupler shafts 132d are formed separately and arranged alternately in the support pipe 130 with the ends contacting each other to form the control shaft 132.

The engaging portions 132c and the coupler shafts 132d have the same diameter, but the engaging portions 132c are shorter than the coupler shafts 132d. The engaging portions 132c are made of a high-strength material. The engaging portions 132c are, for example, made of an iron based material such as cast steel or cast iron. The coupler shafts 132d are made of an aluminum alloy material like the cylinder head 8.

Each coupler shaft 132d is formed as a rod having a circular cross-section. Each engaging portion 132c has a support hole 132b extending in a radial direction as shown in Fig. 10(B). The proximal portion of a control pin 132a is inserted into each support hole 132b, so that the control pin 132a projects in a radial direction. Like the engaging portions 132c, the control pins 132a are made of a high-strength material. The control pins 132a are, for example, made of an iron based material such as cast steel or cast iron.

When the control shaft 132 is provided in the support pipe 130, the distal end of each control pin 132a extends through one of the elliptical holes 130a formed in the support pipe 130. As shown Fig. 12, the control pin 132a fits in the circumferential groove 128g formed in the corresponding slider gear 128.

As shown in Fig. 11, the coupler shaft 132d at the right end of the control shaft 132 receives axial force from the ball screw shaft 100e of the slide actuator 100. The engaging portion 132c at the left end of the control shaft 132 is urged toward the slide actuator 100 by a spring 102a provided in the urging mechanism 102 (Fig. 2) through an auxiliary shaft 133 and a pressing shaft 102c provided on a spring seat 102b.

When transmitting force for driving the intake cams 45a, the four intervening drive mechanisms 120 apply an axial force to the control shaft 132 toward the urging mechanism 102 through the control pins 132a by means of the inner spline mechanism. The force of the spring 102a is slightly greater than the axial force generated by the four intervening drive mechanisms 120.

Therefore, when moving the entire control shaft 132 toward the urging mechanism 102 (in a direction indicated by arrow L), the slide actuator 100 moves the ball screw shaft 100e against a force that is equivalent to the difference between the force of the spring 102a and the force generated by the intervening drive mechanisms 120. When moving the control shaft 132 in the opposite direction (in a direction indicated by arrow H), the slide actuator 100 weakens a force of the ball screw shaft 100e that acts against the force of the spring 102a, so that a force in the opposite direction is generated. Accordingly, the control shaft 132 is moved using the urging force of the spring 102a.

In this manner, when the ball screw shaft 100e is moved toward the urging mechanism 102 (in the direction of arrow L), the force of the urging mechanism 102 applied through the auxiliary shaft 133 moves the four engaging portions 132c and the four coupler shafts 132d toward the urging mechanism 102

while maintaining the contacting state. Accordingly, in the inner spaces of the intervening drive mechanisms 120, which are defined by the input portions 122 and the rocker cams 124, 126, all the slider gears 128 engaged with the control pins 132a of the engaging portions 132c are moved in the direction of arrow L (Fig. 4) by the same amount as the displacement amount of the ball screw shaft 100e.

When the ball screw shaft 100e is moved in the same direction as the direction of the urging force of the spring 102a by weakening the force acting against the urging force of the spring 102a or by generating a force in the opposite direction, the slider gears 128 are also moved by the same amount as the displacement amount of the ball screw shaft 100e. That is, the four engaging portions 132c and the four coupler shafts 132d are moved toward the slide actuator 100 while maintaining the contacting state. Accordingly, in the inner spaces of the intervening drive mechanisms 120, all the slider gears 128 engaged with the control pins 132a of the engaging portions 132c are moved in the direction of arrow H (Fig. 4) by the same amount as the displacement amount of the ball screw shaft 100e.

Therefore, although the control shaft 132 is formed with the four engaging portions 132c and the four coupler shafts 132d, which are formed separately with end faces contacting each other in the support pipe 130, the control shaft 132 is capable of moving the slider gears 128 by the same amount.

In this manner, the axial position of each slider gear 128 is determined by displacement amount of the control shaft 132. However, since each slider gear 128 is engaged with the corresponding control pin 132a at the circumferential groove 128g, each slider gear 128 can be movable along the circumferential direction regardless of the position of the

corresponding control pin 132a.

In each slider gear 128, the input helical spline 128a is meshed with the helical spline 122b in the input portion 122. The first output helical spline 128c is meshed with the helical spline 124b in the first rocker cam 124, and the second output helical spline 128e is meshed with the helical spline 126b in the second rocker cam 126.

Each intervening drive mechanism 120 contacts bearings 136 provided on the cylinder head 8 at the bearings 124c, 126c of the rocker cams 124, 126. Each intervening drive mechanism 120 is movable about the axis but prevented from moving along the axis by the corresponding bearings 136. Therefore, even if the control shaft 132 axially moves the slider gears 128, the input portions 122 and the rocker cams 124, 126 are not moved along the axis.

Thus, by adjusting the amount of axial displacement of the slider gears 128 in the inner space of the intervening drive mechanisms 120, the phase difference between each input portion 122 and the corresponding rocker cams 124, 126 can be changed by means of the helical splines 128a, 122b, 128c, 124b, 128e, 126b. Accordingly, the position of each roller 122f relative to the corresponding noses 124d, 126d is changed.

The variable valve actuation mechanism is assembled in the following manner. The coupler shafts 132d and the engaging portions 132c are inserted into the support pipe 130 while arranged alternately as shown in Fig. 11. Then, as shown in Fig. 5, the support pipe 130 is inserted into the through holes 128f of the four slider gears 128 such that each slider gear 128 is located at a position corresponding to one of the elliptical holes 130a. The proximal end of the

control pin 132a is inserted into the pin insertion hole 128h of each slider gear 128 through the corresponding elliptical hole 130a. Subsequently, the integrated support pipe 130 and the control shaft 132 are rotated relative to the slider gears 128, so that each control pin 132a is sufficiently spaced from the corresponding pin insertion hole 128h. This prevents each control pin 132a from falling off the corresponding support hole 132b even if the slider gears 128 is moved relative to the integrated support pipe 130 and the control shaft 132. Thereafter, the input portions 122 and the rocker cams 124, 126 are assembled with the slider gears 128 to form an assembly.

The assembly is then provided on and fixed to the cylinder head 8 as shown in Fig. 2. The slide actuator 100 is attached to the right end of the control shaft 132. The auxiliary shaft 133 shown in Fig. 11 is provided at the left end of the control shaft 132, and the urging mechanism 102 is attached to the bearings 136 to press the auxiliary shaft 133 with a pressing shaft 102c. This configuration permits the position of each roller 122f relative to the corresponding noses 124d, 126d to be changed by the slide actuator 100, thereby adjusting the valve lift of the intake valves 12.

Figs. 13(A) and 13(B) show states of one of the intervening drive mechanisms 120, in which the force of the slide actuator 100 is adjusted such that the ball screw shaft 100e moves the control shaft 132 in the direction of arrow L (see Figs. 3 and 4) by the maximum amount against the force of the urging mechanism 102. Fig. 13(A) shows a valve closing state, and Fig. 13(B) shows a valve opening state. In these states, the roller 122f of the input portion 122 and the noses 124d, 126d of the rocker cams 124, 126 are closest to each other. Therefore, even if the intake cam 45a presses the roller 122f of the input portion 122 to the lowest position

as shown in Fig. 13(B), the cam surfaces 124e, 126e of the noses 124d, 126d press the rocker roller 52a by the minimum amount, which minimizes the valve lift of the intake valve 12. Accordingly, the intake air amount from the intake ports 14 into the combustion chamber 10 is minimized.

Figs. 14(A) and 14(B) show states of one of the intervening drive mechanisms 120, in which the force of the slide actuator 100 is adjusted such that the ball screw shaft 100e moves in the same direction as the force of the urging mechanism 102, and the force of the urging mechanism 102 is used to move the control shaft 132 in the direction of arrow H (see Figs. 3 and 4) by the maximum amount. Fig. 14(A) shows a valve closing state, and Fig. 14(B) shows a valve opening state. In these states, the roller 122f of the input portion 122 and the noses 124d, 126d of the rocker cams 124, 126 are furthest from each other. Therefore, even if the intake cam 45a presses the roller 122f of the input portion 122 to the lowest position as shown in Fig. 14(B), the cam surfaces 124e, 126e of the noses 124d, 126d press the rocker roller 52a by the maximum amount, which maximizes the valve lift of the intake valve 12. Accordingly, the intake air amount from the intake ports 14 into the combustion chamber 10 is maximized.

In this manner, by adjusting the axial position of the control shaft 132 by the slide actuator 100 and the urging mechanism 102, the valve lift of the intake valves 12 is continuously varied between the state of Figs. 13(A), 13(B) and the state of Figs. 14(A), 14(B). This permits the intake air amount to be adjusted without using a throttle valve.

When adjusting the axial position of the control shaft 132, each control pin 132a alternately receives reaction forces indicated by arrows shown in the cross-sectional view of Fig. 15 from the circumferential groove 128g of the

corresponding slider gear 128, or alternately receives a strong reaction force and a weak reaction force. Therefore, the control pins 132a and the engaging portions 132c, in which the support holes 132b are formed, are made of an iron based material. The coupler shafts 132d, which do not directly receive the reaction forces, are made of an aluminum alloy material. In this embodiment, the auxiliary shaft 133 is made of an aluminum alloy material like the coupler shafts 132d. However, the auxiliary shaft 133 may be made of an iron based material.

In this embodiment, when the valve lift of the intake valves 12 is minimized, the intake valves 12 are opened by a small amount as shown in Fig. 13(B). However, the valve lift may be set to zero, that is, the intake valves 12 may be completely closed. In this case, the intake air amount is zero.

In the above described configuration, the slider gears 128 correspond to valve actuation controllers, the slide actuator 100 corresponds to an actuator, and the urging mechanisms 102 correspond to urging means. The rocker cams 124, 126 correspond to output portions, and the combination of the helical spline 122b of each input portion 122 and the input helical spline 128a of the corresponding slider gear 128 corresponds to a first spline mechanism. The combination of the helical splines 124b, 126b of each set of the rocker cams 124, 126, and the helical splines 128c, 128e of the corresponding slider gear 128 corresponds to a second spline mechanism. The first spline mechanism is of a right-hand screw type, and the second spline mechanism is of a left-hand screw type. The helix angles of the first and second spline mechanisms thus different.

The first embodiment described above has the following

advantages.

(1) In the control shaft 132, the engaging portions 132c are engaged with the slider gears 128 by the control pins 132a, and are made of a high-strength iron based material as described above. The remaining portions of the control shaft 132, that is, the coupler shafts 132d do not need to be as strong as the engaging portions 132c. The coupler shafts 132d are therefore made of an aluminum alloy material like the cylinder head 8. Therefore, compared to a case where the control shaft 132 is entirely made of an iron based material, the thermal expansion coefficient of the control shaft 132 is closer to that of the cylinder head 8. Specifically, the coefficient of linear expansion of an iron based material is approximately 10 to 12×10^{-6} (1/°C), while the coefficient of linear expansion of an aluminum alloy material is 24 to 25×10^{-6} (1/°C). Therefore, compared to a case where the control shaft 132 is made only of an iron based material, the thermal expansion coefficient of the control shaft 132 is made closer to that of the cylinder head 8.

Thus, even if the ambient temperature changes, the relationship between the spaces among components of the cylinder head 8 (in this embodiment, the spaces between the bearings 136 and the slide actuator 100) and the distance between the slide actuator 100 and each engaging portion 132c of the control shaft 132 is hardly changed. That is, the position of each engaging portion 132c relative to the cylinder head 8 is prevented from being displaced.

In this embodiment, the axial length of each engaging portion 132c is sufficiently shorter than that of each coupler shaft 132d, and is about 10 to 20% of the entire length of the control shaft 132. Therefore, the thermal expansion coefficient of the entire control shaft 132 is not

significantly different from the case where the entire control shaft 132 is made of an aluminum alloy material. The control shaft 132 thus receives little influence from changes in the ambient temperature.

Also, since the engaging portions 132c of the control shaft 132 are made of an iron based material, the engaging portions 132c have a sufficient strength, which prevents the control shaft 132 from being deformed at the engaging portions 132c.

Accordingly, the variable valve actuation mechanism of this embodiment is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft 132.

(2) At the end opposite to the end where the slide actuator 100 is provided, the urging mechanism 102 urges the control shaft 132 toward the slide actuator 100 through the auxiliary shaft 133. Therefore, the engaging portions 132c and the coupler shafts 132d of the control shaft 132 do not need to be integrated. That is, although formed separately, the engaging portions 132c and the coupler shafts 132d contact each other at end faces. Thus, the slide actuator 100 axially moves the control shaft 132 while maintaining the contacting states of the engaging portions 132c and the coupler shafts 132d. Since the engaging portions 132c and the coupler shafts 132d, which are made of different materials, do not need to be connected, the manufacture of the control shaft 132 is simplified.

The engaging portions 132c and the coupler shafts 132d are shaped as rods having a circular cross-section and have flat end faces. Further, each engaging portion 132c has a simple shape only with the circular support hole 132b. By

accurately forming the engaging portions 132c and the coupler shafts 132d into predetermined lengths, the position of each control pin 132a is accurately determined. The manufacture of the control shaft 132 is therefore simple.

Further, when adjusting the length of the control shaft 132 or when changing the length of the control shaft 132 according to the type of the engine, the coupler shafts 132d are only replaced by coupler shafts 132d having a different length, and the common engaging portions 132c can be used. This reduces the costs of the control shaft 132.

(3) The spring 102a of the urging mechanism 102 applies a force to the control shaft 132 against the axial force generated by the four intervening drive mechanisms 120. The force applied by the urging mechanism 102 is slightly greater than the axial force generated by the four intervening drive mechanisms 120. Therefore, although generating a relative small output, the slide actuator 100 is capable of adjusting the axial position of the control shaft 132, while maintaining the contacting state of the engaging portions 132c and the coupler shafts 132d.

Therefore, energy consumption during operation of the variable valve actuation mechanism is reduced, and a compact motor 100a can be used. This reduces the size and the weight of the engine 2.

[Second embodiment]

A control shaft 232 of the second embodiment has four engaging portions 232c-1 to 232c-4 and four coupler shafts 232d-1 to 232d-4. The coupler shafts 232d-1 to 232d-4 are made of a material having a thermal expansion coefficient that is greater than that of the cylinder head 8. The coupler

shafts 232d-1 to 232d-4 are used with the engaging portions 232c-1 to 232c-4, which are made of an iron based material, such that the thermal expansion coefficient of the entire control shaft 232 is substantially equal to that of the cylinder head 8. Control pins 232a-1 to 232a-4 are made of an iron-based material. The other components are the same as those in the first embodiment, and are thus given the same reference numerals as the first embodiment.

The engaging portions 232c-1 to 232c-4 are made of an iron-based material the linear expansion coefficient of which is represented by **a** (1/°C). The coupler shafts 232d-1 to 232d-4 are made of an aluminum alloy material the linear expansion coefficient of which is represented by **b** (1/°C). The cylinder head 8 is made of an aluminum alloy material the linear expansion coefficient of which is represented by **c** (1/°C). The material for the coupler shafts 2323-1 to 232d-4 is selected such that an inequality $\mathbf{a} < \mathbf{c} < \mathbf{b}$ is satisfied.

In this case, a resultant linear expansion coefficient **d1** at a position corresponding to the control pin 232a-1 that is closest to the slide actuator 100 having the shaft position sensor 100d changes as indicated by a solid line in the graph of Fig. 17.

As shown in Fig. 16, the length of the coupler shaft 232d-1 is represented by x1, the length of the engaging portion 232c-1 is represented by y1. The length from the end face of the engaging portion 232c-1 contacting the coupler shaft 232d-1 to the control pin 232a-1 is represented by y1/2. In this case, the equation RX:RY = x1:(y1/2) is satisfied.

Therefore, by adjusting the ratio x1:(y1/2), the linear expansion coefficient d1 is made to match the linear expansion coefficient **c** of the cylinder head 8.

When expressed with the linear expansion coefficients **a**, **b**, and **c**, the value RX/RY is (**c** - **a**)/(**b** - **c**). The following equation (1) is therefore satisfied.

$$(\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c}) = x1/(y1/2) \qquad (1)$$

According to the equation (1), x1 is expressed by the following equation (2).

$$x1 = y1 \times (\mathbf{c} - \mathbf{a})/2 \times (\mathbf{b} - \mathbf{c}) \qquad (2)$$

The value (x1 + y1/2), which represents the distance between the slide actuator 100 and the intervening drive mechanism 120 closest to the slide actuator 100 is represented by L, y1 is expressed by the following equation (3).

$$y1 = 2(L - x1) \qquad (3)$$

Substituting the right side of the equation (3) in y1 of the equation (2) yields the value of x1 expressed by the equation (4). Accordingly, y1 is determined as expressed by the equation (5).

$$x1 = L(\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{a}) \qquad (4)$$

$$y1 = 2L(\mathbf{b} - \mathbf{c})/(\mathbf{b} - \mathbf{a}) \qquad (5)$$

Likewise, a resultant linear expansion coefficient d2 at a position corresponding to the second control pin 232a-2 from the slide actuator 100 changes as shown in Fig. 17. The length of the coupler shaft 232d-2 is represented by x2, and the length of the engaging portion 232c-2 is represented by y2. The length from the end face of the engaging portion 232c-2 contacting the coupler shaft 232d-2 to the control pin 232a-2 is represented by y2/2. In this case, the equation

$RX:RY = x1 + x2:(y1+y2/2)$ is satisfied.

Therefore, the following equation (6) is satisfied.

$$(\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c}) = (x1 + x2)/(y1 + y2/2) \qquad (6)$$

According to the equation (6), x2 is expressed by the following equation (7).

$$x2 = [(y1 + y2/2) \times (\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c})] - x1 \qquad (7)$$

The values of x1 and y1 are determined by the equations (4) and (5). The value (y1/2 + x2 + y2/2), which represents the distance between each adjacent pair of the intervening drive mechanisms 120 is represented by M, y2 is expressed by the following equation (8).

$$y2 = 2(M - y1/2 - x2) \qquad (8)$$

Substituting the right side of the equation (8) in y2 of the equation (7) yields the value of x1 expressed by the equation (9).

$$x2 = [(y1/2 + M)(\mathbf{c} - \mathbf{a}) - x1(\mathbf{b} - \mathbf{c})]/(\mathbf{b} - \mathbf{a}) \qquad (9)$$

Since x2 is determined, y2 is determined using the equation (8).

Likewise, a resultant linear expansion coefficient d3 at a position corresponding to the third control pin 232a-3 from the slide actuator 100 changes as shown in Fig. 17. The length of the coupler shaft 232d-3 is represented by x3, and the length of the engaging portion 232c-3 is represented by y3. The length from the end face of the engaging portion 232c-3 contacting the coupler shaft 232d-3 to the control pin

232a-3 is represented by y3/2. In this case, the equation RX:RY = x1 + x2 + x3:(y1 + y2 + y3/2) is satisfied.

Therefore, the following equation (10) is satisfied.

$$(\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c}) = (x1 + x2 + x3)/(y1 + y2 + y3/2) \quad (10)$$

According to the equation (10), x3 is expressed by the following equation (11).

$$x3 = [(y1 + y2 + y3/2) \times (\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c})] - (x1 + x2) \quad (11)$$

The values of x1, x2, y1 and y2 are determined by the equations (4), (5), (8), and (9). The value (y2/2 + x3 + y3/2), which represents the distance between each adjacent pair of the intervening drive mechanisms 120, is represented by M, y3 is expressed by the following equation (12).

$$y3 = 2(M - y2/2 - x3) \quad (12)$$

Substituting the right side of the equation (12) in y3 of the equation (11) yields the value of x3 expressed by the equation (13).

$$x3 = [(y1 + y2/2 + M)(\mathbf{c} - \mathbf{a}) - (x1 + x2)(\mathbf{b} - \mathbf{c})]/(\mathbf{b} - \mathbf{a}) \quad (13)$$

Since x3 is determined, y3 is determined using the equation (12).

Likewise, a resultant linear expansion coefficient d4 at a position corresponding to the fourth control pin 232a-4 from the slide actuator 100 changes as shown in Fig. 17. The length of the coupler shaft 232d-4 is represented by x4, and

the length of the engaging portion 232c-4 is represented by y4. The length from the end face of the engaging portion 232c-4 contacting the coupler shaft 232d-4 to the control pin 232a-4 is represented by y4/2. In this case, the equation RX:RY = x1 + x2 + x3 + x4:(y1 + y2 + y3 + y4/2) is satisfied.

Therefore, the following equation (14) is satisfied.

$$(\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c}) = (x1 + x2 + x3 + x4)/(y1 + y2 + y3 + y4/2) \quad (14)$$

According to the equation (14), x4 is expressed by the following equation (15).

$$x4 = [(y1 + y2 + y3 + y4/2) \times (\mathbf{c} - \mathbf{a})/(\mathbf{b} - \mathbf{c})] - (x1 + x2 + x3) \quad (15)$$

The values of x1, x2, x3, y1, y2, and y3 are determined by the equations (4), (5), (8), (9), (12), and (13). The value (y3/2 + x4 + y4/2), which represents the distance between each adjacent pair of the intervening drive mechanisms 120, is represented by M, y4 is expressed by the following equation (16).

$$y4 = 2(M - y3/2 - x4) \quad (16)$$

Substituting the right side of the equation (16) in y4 of the equation (15) yields the value of x4 expressed by the equation (17).

$$x4 = [(y1 + y2 + y3/2 + M)(\mathbf{c} - \mathbf{a}) - (x1 + x2 + x3)(\mathbf{b} - \mathbf{c})]/(\mathbf{b} - \mathbf{a}) \quad (17)$$

Since x4 is determined, y4 is determined using the equation (16).

In this manner, all the lengths x1 to x4 and y1 to y4 of the coupler shafts 232d-1 to 232d-4 and the engaging portions 232c-1 to 232c-4 are determined. Then, the coupler shafts 232d-1 to 232d-4 and the engaging portions 232c-1 to 232c-4 having the lengths x1 to x4 and y1 to y4 are formed and assembled as the control shaft 232 of the variable valve actuation mechanism as described in the first embodiment. As a result, the thermal expansion coefficient of the cylinder head 8 is substantially the same as the thermal expansion coefficient of the control shaft 232. Thus, even if the ambient temperature changes, the accuracy of the control of the intervening drive mechanisms 120 are not affected.

The second embodiment as described above has the following advantages.

(1) Since the material and lengths of the engaging portions 232c-1 to 232c-4, and the material and the lengths of the coupler shafts 232d-1 to 232d-4 are determined as described above, the thermal expansion coefficient of the entire control shaft 232 is substantially the same as the thermal expansion coefficient of the cylinder head 8. Therefore, even if the ambient temperature changes, the positions of the engaging portions 232c-1 to 232c-4 relative to the cylinder head 8 are not displaced.

Also, since the engaging portions 232c-1 to 232c-4 are made of an iron based material, the engaging portions 232c-1 to 232c-4 have a sufficient strength, which prevents the control shaft 232 from being deformed at the engaging portions 232c-1 to 232c-4.

Accordingly, the variable valve actuation mechanism of this embodiment is capable of performing accurate valve

actuation control, while maintaining the strength of the control shaft 232.

(2) The same advantages as the items (2) and (3) of the first embodiment are provided.

[Third embodiment]

In a third embodiment, a control shaft 282 includes a single shaft main body 282d as shown in Fig. 18(A). The shaft main body 282d is made of the same material as that of the cylinder head or of an aluminum alloy material the thermal expansion coefficient is substantially the same as that of the material of the cylinder head.

As shown in Fig. 18(B), the shaft main body 282d has rectangular holes 282e, the number of which is the same as the number of the cylinders. Each rectangular hole 282e extends in the axial direction. An engaging portion 282c, the shape of which is substantially the same as the rectangular holes 282e, is fitted in each rectangular hole 282e.

A support hole 282b is formed in each engaging portion 282c. As shown in Fig. 19, the proximal end of a control pin 282a is inserted into and supported by each support hole 282b. The engaging portions 282c and the control pins 282a are both made of a high-strength iron based material.

The control shaft 282 is entirely accommodated in a support pipe described in the first embodiment. Each engaging portion 282c is engaged with the slider gear of the corresponding intervening drive mechanism described in the first embodiment by the control pin 282a. As shown in Fig. 11, the slide actuator and the urging mechanism are located at the ends of the control shaft 282, so that the valve

actuation of the intake valves is changed as described in the first embodiment. The other configurations are the same as those of the first embodiment.

The third embodiment described above has the following advantages.

(1) The engaging portions 282c are engaged with the slide gears by means of the control pins 282a. Also, each engaging portion 282c is provided only about the corresponding control pin 282a to support the control pin 282a.

Therefore, the shaft main body 282d, which corresponds to the remaining portions of the control shaft 282, is formed integrally while fitting the engaging portions 282c in the rectangular holes 282e. Thus, the shaft main body 282d of the control shaft 282 is formed of a material that is continuous along the entire axial length.

Since the shaft main body 282d does not need to have a high strength as the engaging portions 282c, the shaft main body 282d may be made of the same material as that of the cylinder head or of an aluminum alloy material the thermal expansion coefficient is substantially the same as that of the material of the cylinder head. The thermal expansion coefficient of the shaft main body 282d is dominant in the entire control shaft 282, which permits the thermal expansion coefficient of the control shaft 282 to be substantially the same as that of the cylinder head.

Therefore, even if the ambient temperature changes, the position of each engaging portion 282c relative to the cylinder head is prevented from being displaced. Also, since the engaging portions 282c are made of an iron based material,

the engaging portions 282c have a sufficient strength, which prevents the control shaft 282 from being deformed at the engaging portions 282c.

Accordingly, the variable valve actuation mechanism of this embodiment is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft 282.

Thus, the item (1) of the advantages of the first embodiment is further enhanced.

(2) The same advantage as the item (3) of the first embodiment is provided.

[Fourth embodiment]

In the fourth embodiment, a control shaft 302 is a rod made of an iron-based material as shown in the perspective view of Fig. 20. A ball screw shaft 302b is integrally formed with the control shaft 302. Thus, the control shaft 302 has no problems related to the strength. Control pins 302a are also made of an iron-based material. The cylinder head of the engine is made of an aluminum alloy. As in the other embodiments, the cylinder head has a cam carrier.

The perspective view of Fig. 21 illustrates the control shaft 302 assembled with a variable valve actuation mechanism. In the example of Fig. 21, intervening drive mechanisms 320 substantially have the same configuration as those in the first embodiment, but arranged in the opposite orientation. Therefore, the intervening drive mechanisms 320 are rocked by intake cams 345a in the direction opposite to the rocking direction of the first embodiment. However, the valves are actuated in the same manner by rocking. To illustrate the

opening state of intake valves 312, valve seats 314 are illustrated as annular bodies corresponding to the intake valves 312.

In this configuration also, when the ball screw shaft 302b is moved in the direction of arrow H by a slide actuator, each nose 326d separates from the corresponding roller 322f, so that the valve actuation value of the intake valves 312 (both of the valve duration angle and the valve lift) are increased. When the ball screw shaft 302b is moved in the direction of arrow L, each nose 326d approaches the corresponding roller 322f, which decreases the valve duration angle and the valve lift of the intake valves 312.

In this embodiment, the leak down properties of lash adjusters 350a, 350b, 350c, and 350d that adjust the valve clearances of the intake valves 312 are different from one cylinder to another. The leak down property refers to a property in which when receiving a certain load, a plunger of each lash adjuster 350a to 350d is moved into the main body of the adjuster due to hydraulic oil leakage inside the lash adjuster 350a to 350d. The greater the leak down property value is, the greater the amount by which the plungers of the lash adjusters 350a to 350d are moved downward relative to the roller rocker arms 352 becomes. The amount of the downward movement will hereinafter be referred to as leak down amount. Therefore, even if the control shaft 302 is not axially moved, the valve duration angle and the valve lift of the intake valves 312 are reduced as the leak down property values are increased.

The leak down property values of the lash adjusters 350a to 350d are set such that the leak down property value LD1 of the lash adjuster 350a of the first cylinder (denoted as #1) is the greatest. The leak down property values LD2, LD3, and

LD4 of the lash adjusters 350b, 350c, and 350d of second cylinder #2, the third cylinder #3, and the fourth cylinder #4 are set to decrease in this order.

That is, at the cylinder #1, the amount of decrease in the valve duration angle and lift due to leak down is the greatest in relation to the amount of adjustment of the control shaft 302. The amount of decrease decreases in the order of the cylinders #2, #3, and #4.

The axial position of an end face 321 of each intervening drive mechanism 320 at the side closer to the ball screw shaft 302b is adjusted, for example, with a shim, at room temperature (for example, 20°C), such that the valve duration angle and lift of the intake valves 312 are constant and not varied among all the cylinders.

Therefore, when the engine is cold, the increase amounts of the valve duration angle and lift due to the difference of thermal expansion coefficient between the cylinder head and the control shaft 302 are zero and the same for all the cylinders. Also, when the engine is cold, hydraulic oil supplied from a hydraulic pump to the lash adjusters 350a to 350d has a high viscosity due to a low temperature. Therefore, although the leak down property values varies as shown in Fig. 22, the amount of decrease in the valve duration angle and lift due to the actual leak down do not vary significantly among the cylinders #1 to #4 as shown in Fig. 23(B). Therefore, as shown in Fig. 24, the relationship between the adjustment amount of the slide actuator and the valve duration angle and lift of each cylinder do not vary significantly.

Therefore, when the engine has been warmed up, the increase amount of the valve duration angle and lift due to

the difference of thermal expansion coefficient between the cylinder head and the control shaft 302 increases in the order of the cylinders #4, #3, #2, and #1. That is, the increase amount significantly varies between the cylinder #1 and the cylinder #4. Further, when the engine has been warmed up, hydraulic oil supplied to the lash adjusters 350a to 350d has a low viscosity due to a high temperature. Therefore, as shown in Fig. 22, the sensitivity of the leak down property is noticeable, and the decrease amount of the valve duration angle and lift due to the actual leak down increases in the order of the cylinders #4, #3, #2, and #1, and significantly vary between the cylinder #1 and the cylinder #4. In this embodiment, the amounts of increase in the valve duration angle and lift due to a difference in the thermal expansion coefficient generated after the engine is warmed up, and the amounts of decrease in the valve duration angle and lift due to leak down cancel each other completely. The increase or decrease in the valve duration angle and lift are therefore set to zero. Therefore, as shown in Fig. 26, the relationship between the adjustment amount of the slide actuator and the valve duration angle and lift of the cylinders completely coincide and do not vary.

Fig. 27 shows a comparison example, in which lash adjusters 350a to 350d have the same leak down property. In this case, after the engine is warmed up, the relationship between the adjustment amount of the slide actuator and the valve duration angle and lift of the cylinders significantly vary as shown in Fig. 27. If the relationships are attempted to be made coincide after the engine is warmed up in the case of the comparison example, where all the lash adjusters have the same leak down property, the relationships will greatly vary in a state where the engine is cold. That is, the relationships cannot be made to coincide both in a period when the engine is cold and in a period in the engine has

been warmed up.

The fourth embodiment described above has the following advantages.

(1) If the leak down property value of the lash adjusters 350a to 350d is small, the valve duration angle and lift corresponding to the amount of valve actuation transmitted from the intervening drive mechanisms 320 are great. As the leak down property value increases, the valve duration angle and lift for the same valve actuation amount are reduced.

The sensitivity to the valve actuation changes due to the leak down properties is related to the viscosity of hydraulic oil. That is, if the viscosity of the hydraulic oil is high, the amount of oil leakage in the lash adjusters 350a to 350d is reduced, which lowers the sensitivity of valve actuation changes due to the leak down properties. That is, even if the leak down properties greatly vary among the lash adjusters 350a to 350d, a high viscosity of the hydraulic oil prevents the difference in the leak down properties from varying the valve duration angle and lift. Also, a low viscosity of the hydraulic oil permits the difference in the leak down properties to vary the valve duration angle and lift by a greater degree.

In this embodiment, the leak down properties are varied among the cylinders to suppress variation of the valve duration angle and lift due to the difference in the thermal expansion coefficient of the control shaft 302 and the cylinder head in relation to the intervening drive mechanisms 320. Therefore, the differences in the thermal expansion coefficient do not cause any problems when the engine has been warmed up.

Further, when the engine is cold, that is, when there is no variation in the valve duration angle and lift due to differences in the thermal expansion coefficient, the hydraulic oil has a high viscosity due to a low temperature. Therefore, even if the leak down properties vary, the valve duration angle and lift are hardly varied by the differences in the leak down properties.

Therefore, even if the thermal expansion coefficients are different between the control shaft 302 and the cylinder head, the valve duration angle and lift are prevented from being varied over the entire temperature range of the engine.

Accordingly, the variable valve actuation mechanism of this embodiment is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft 302.

[Fifth embodiment]

The fifth embodiment is different from the fourth embodiment in that the leak down properties of lash adjusters 450a, 450b, 450c, 450d shown in Fig. 28 are all the same. The lash adjusters 450a to 450d, each of which corresponding to one of the cylinders, receive hydraulic oil form a hydraulic pump 464 driven by the engine through hydraulic pressure adjuster mechanisms 448a, 448b, 448c, and 448d, respectively. The other configurations are substantially the same as those of the fourth embodiment shown in Figs. 20 and 21.

Although the leak down properties are the same for all the lash adjusters 450a to 450d, the amount of decrease in the valve duration angle and lift due to leak down vary according to the pressure of supplied hydraulic oil as shown

in Fig. 29. That is, when the pressure of supplied oil is low, the leak down amount is increased. Accordingly, the amount of decrease in the valve duration angle and lift is increased. When the pressure of supplied oil is high, the leak down amount is decreased. Accordingly, the amount of decrease is decreased.

An ECU 460 utilizes the phenomenon shown in Fig. 29 to compute the pressure of hydraulic oil supplied to the lash adjusters 450a to 450d based on the engine temperature (in this embodiment, the engine coolant temperature THW detected by the coolant temperature sensor 462), by referring to an oil pressure control map shown in Fig. 30.

For example, when the engine is cold, a common pressure value (P4) for supplying hydraulic oil is obtained for all the cylinders #1 to #4 using the oil pressure control map (Fig. 30). Accordingly, the ECU 460 outputs a control signal to all the hydraulic pressure adjuster mechanisms 448a to 448d such that the pressure of oil supplied from the hydraulic pump 464 to the lash adjusters 450a to 450d seek the oil pressure P4.

As the engine coolant temperature THW increases, the supplied oil pressure value obtained from the oil pressure control map (Fig. 30) varies among the cylinders. The value of pressure of oil supplied to the lash adjuster 450d corresponding to the cylinder #4 closest to the slide actuator (to the ball screw shaft 402b) is the highest. The supplied oil pressure value decreases in the order of the lash adjuster 450c corresponding to the cylinder #3, the lash adjuster 450b corresponding the cylinder #2, and the lash adjuster 450a corresponding to the cylinder #1. After the engine is warmed up, the supplied oil pressure value corresponding to the cylinder #1 is set to a value P1, the

supplied oil pressure value corresponding to the cylinder #2 is set to a value P2, the supplied oil pressure value corresponding to the cylinder #3 is set to a value P3, and the supplied oil pressure value corresponding to the cylinder #4 is set a value P4. The inequality P1 < P2 < P3 < P4 is satisfied.

The relationship of the supplied oil pressure values P1 to P4 is determined such that the amount of increase in the valve duration angle and lift by the intervening drive mechanisms 420 due to the difference in the thermal expansion coefficient between the cylinder head and a control shaft 402 is cancelled by the amount of decrease in the valve duration angle and lift due to leak down.

The fifth embodiment described above has the following advantages.

(1) As shown in Fig. 29, increasing the pressure of oil supplied to the lash adjusters 450a to 450d decreases the amount of leak down. Accordingly, the valve duration angle and lift are maintained to be great. Decreasing the supplied oil pressure increases the amount of leak down. Accordingly, the valve duration angle and lift are reduced.

Taking advantages of these facts, the ECU 460 adjusts the pressure of oil supplied to the lash adjusters 450a to 450d according to the engine temperature (in this embodiment, the engine coolant temperature THW), thereby preventing the valve duration angle and lift from being changed by the difference in the thermal expansion coefficients between the cylinder head and the control shaft 402. That is, using the oil pressure control map of Fig. 30, the ECU 460 decreases the pressure of oil supplied to the lash adjusters 450a to 450d according to the engine coolant temperature THW in the

order of the cylinder #4 to the cylinder #1.

In this manner, by adjusting the supplied oil pressure, variations of the valve duration angle and lift due to differences in the thermal expansion coefficient are suppressed over the entire temperature range of the engine.

Accordingly, the variable valve actuation mechanism of this embodiment is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft 402.

[Sixth embodiment]

Fig. 31 illustrates a control shaft 532, which includes engaging portions 532c and coupler shafts 532d. The sixth embodiment is different from the first embodiment in that the relationship in the lengths of each engaging portion 532c and corresponding coupler shaft 532d varies between the intervening drive mechanisms, which are arranged at an equal interval.

That is, in the first embodiment, the engaging portions 132c, which are made of an iron based material such as cast steel or cast iron, are formed to have the minimum length. The coupler shafts 132d, which are made of an aluminum alloy material like the cylinder head, are formed to have the maximum length. Accordingly, the difference of the thermal expansion coefficient between the control shaft and the cylinder head is minimized. Particularly, the coupler shafts 132d of the intervening drive mechanisms have the same length.

In the sixth embodiment, the lengths of the engaging portions 532c and the coupler shafts 532d are determined such that the displacement of the positions of the control pins

532a due to thermal expansion coefficient difference is within the permissible range, and that the lengths of the coupler shafts 532d are minimized.

The slide actuator of the sixth embodiment axially moves and detects the axial position of a ball screw shaft 500e, which has a proximal end 500f. A length relationship R4 between the proximal end 500f of the ball screw shaft 500e and the control pin 532a(#4) corresponding to the fourth cylinder #4 is set as shown in Fig 32(A). That is, the relationship between the length a4 of the coupler shaft 532d(#4) corresponding to the fourth cylinder #4 and the length b4 of the remainder of the ball screw shaft 500e and the engaging portion 532c(#4) corresponding to the fourth cylinder #4 is determined such that a displacement V4 of the control pin 532a(#4) relative to the cylinder head is less than a permissible displacement amount (an alternate long and short dash line) between when the engine is cold and when the engine has been warmed up. In Figs. 32(A) to 32(D), each slant solid line represents a state in which a displacement amount increases as the ratio of the length of the corresponding coupler shaft 532d decreases. In all of Figs. 32(A) to 32(D), the horizontal axis represents the distance from the proximal end 500f of the ball screw shaft 500e. A slant broken line represents a displacement amount in a case where the control shaft is entirely made of an iron based material such as cast steel or cast iron.

In this embodiment, no coupler shaft 532d(#4) may be provided between the ball screw shaft proximal end 500f and the control pin 532a(#4). That is, only the ball screw shaft 500e and the engaging portion 532c(#4) may be provided between the proximal end 500f and the control pin 532a (#4). In other words, the length a4 of the coupler shaft 532d(#4) may be zero (a4 = 0 (mm)).

A length relationship R3 between the control pin 532a(#4) and the control pin 532a(#3) corresponding to the third cylinder #3 is set as shown in Fig 32(B). That is, the relationship between the length a3 of the coupler shaft 532d(#3) corresponding to the third cylinder #3 and the length b3 of the remainder of the engaging portion 532c(#3, #4) corresponding to the third and fourth cylinders #3 and #4 is determined such that a displacement V3 of the control pin 532a(#3) relative to the cylinder head is less than the permissible displacement amount between when the engine is cold and when the engine has been warmed up. The displacement amount V3 contains the displacement amount V4. In this embodiment, the length a3 of the coupler shaft 532d(#3) is longer than the length a4 of the coupler shaft 532d(#4).

Further, a length relationship R2 between the control pin 532a(#3) and the control pin 532a(#2) corresponding to the second cylinder #2 is set as shown in Fig 32(C). That is, the relationship between the length a2 of the coupler shaft 532d(#2) corresponding to the second cylinder #2 and the length b2 of the remainder of the engaging portion 532c(#2, #3) corresponding to the second and third cylinders #2 and #3 is determined such that a displacement V2 of the control pin 532a(#2) relative to the cylinder head is less than the permissible displacement amount between when the engine is cold and when the engine has been warmed up. The displacement amount V2 contains the displacement amount V3, and the length a2 of the coupler shaft 532d(#2) is longer than the length a3 of the coupler shaft 532d(#3).

Further, a length relationship R1 between the control pin 532a(#2) and the control pin 532a(#1) corresponding to the first cylinder #1 is set as shown in Fig 32(D). That is, the relationship between the length a1 of the coupler shaft

532d(#1) corresponding to the first cylinder #1 and the length b1 of the remainder of the engaging portion 532c(#1, #2) corresponding to the first and second cylinders #1 and #2 is determined such that a displacement V1 of the control pin 532a(#1) relative to the cylinder head is less than the permissible displacement amount between when the engine is cold and when the engine has been warmed up. The displacement amount V1 contains the displacement amount V2, and the length a1 of the coupler shaft 532d(#1) is longer than the length a2 of the coupler shaft 532d(#2).

No coupler shaft made of an aluminum alloy material is provided between the control pin 532a(#1) and an urging mechanism 502 since this section is not pertinent to displacement amount.

The sixth embodiment described above has the following advantages.

(A) The material of the coupler shafts 532d(#1 to #4) is the same as that of the cylinder head. The material of the engaging portions 532c(#1 to #4) is an iron based material such as cast steel or cast iron, which has a smaller thermal expansion coefficient and a higher strength than that of the cylinder head. The lengths relationships between the engaging portions 532c(#1 to #4) and the coupler shafts 532d(#1 to #4) are determined such that the displacement (V1 to V4) in each position does not exceed the permissible range and that the ratio of the coupler shaft 532d(#1 to #4) increases as the distance from the slide actuator increases. That is, the ratio of the lengths of the coupler shafts 532d(#1 to #4) to the engaging portions 532c(#1 to #4) gradually increases as the distance from the slide actuator increases. Particularly, among the intervening drive mechanisms, the coupler shafts 532d(#1 to #3) is elongated as the distance from the slide

actuator is increased (a3 < a2 < a1).

The material used for the coupler shafts 532d(#1 to #4) corresponding to the other portions in claims is selected for a great thermal expansion coefficient, and therefore, in some cases, cannot be elongated due to an expected reduction of strength of expected increase in the cost. In this embodiment, since the coupler shafts 532d are made of the same aluminum alloy material as the cylinder head, the cost is inevitably increased. Also, since the diameter of the coupler shafts 532d is limited, the strength of the coupler shafts 532d cannot be significantly increased.

Therefore, instead of making the thermal expansion coefficient of the control shafts 532 to be close to that of the cylinder head, it is advantageous for reducing the cost and increasing the strength to set a permissible range of axial displacement of each control pin 532a and to minimize the length of the coupler shafts 532d.

However, in a portion close to the slide actuator, that is, in a case where the coupler shaft 532d(#4 or #3) is shortened in a portion close to the ball screw shaft 500e, the axial displacement of the engaging portions 532c at the control pins 532a accumulates as the distance from the slide actuator increases. Therefore, if the ratio of the length of the coupler shaft 532d to that of the engaging portion 532c is constant regardless of the distance from the slide actuator, the axial displacement of the engaging portions 532c possibly exceeds the permissible range.

In this embodiment, as described above, the ratio of the length of the coupler shafts 532d(#1 to #3) to the length of the engaging portions 532c(#1 to #3) gradually increases as the distance from the slide actuator increases. The length

relationship between the coupler shaft 532d(#4) and the engaging portion 532c(#4) is also increased as the distance from the slide actuator increases.

Therefore, at each engaging portion 532c, the axial displacement of the control pin 532a is prevented from exceeding the permissible range. Also, the strength of the control shaft 532 is prevented from deteriorating, and the cost of the control shaft 532 is prevented from being increased.

Accordingly, the variable valve actuation mechanism of this embodiment is capable of performing accurate valve actuation control, while maintaining the strength of the control shaft 532.

(B) The same advantages as the items (2) and (3) of the first embodiment are provided.

[Other Embodiments]

(a) In the first and second embodiment, the control shaft is configured such that the engaging portions and the coupler shafts are separately formed and contact each other at end faces. The configuration may be changed as shown in Fig. 33. That is, a male thread portion 633a and a female thread portion 633b may be formed in each engaging portion 632c and each coupler shaft 632d, respectively, and the engaging portions 632c and the coupler shafts 632d may be integrated by threading. Alternatively, the engaging portions and the coupler shafts may be integrated by any other means.

Further, if the control shaft is integrated with the ball screw shaft 100e of the slide actuator 100, the axial position of the control shaft can be adjusted only by the

slide actuator without using the urging mechanism 102. Even if the engaging portions, the coupler shafts, and the ball screw shaft are integrated, the urging mechanism 102 may be used to assist the movement of the control shaft.

(b) In the illustrated embodiment, an aluminum alloy material is used as a light ally material. However, a magnesium alloy material may be used.

(c) The slide actuator 100 is a combination of the electric motor and the ball screw. However, a slide actuator having a hydraulic drive source may be used.

(d) In the third embodiment, the shaft main body 282d is an integral body forming the entire control shaft 282. However, as shown in Fig. 11, the control shaft 282 may be separately formed to correspond to each of the cylinders. Alternatively, the control shaft 282 may be separately formed of sections that are integrated by threading. Alternatively, the engaging portions and the coupler shafts may be integrated by any other means.

Further, unlike the rectangular holes shown in Fig. 18, holes having other shapes may be formed to receive the engaging portions.

(e) In the sixth embodiment, the coupler shafts are made of the same material as that of the cylinder head. However, the coupler shafts may be made of a material that is different from that of the cylinder head and has a greater thermal expansion coefficient than that of the engaging portions.